15th Australasian Fluid Mechanics Conference
The University of Sydney, Sydney, Australia
13-17 December 2004

Influence of Free-Stream Turbulence on Wake-Wake Interaction in an Axial Compressor

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Abstract

A turbulence generating grid has been used to increase the level of free-stream turbulence inside a 1.5 stage axial compressor to values typical of an embedded stage in a multi-stage machine. Hot-wire measurements taken in the rotor-stator axial gap have been ensemble-averaged to determine periodic fluctuations in turbulence level and velocity. These results are compared to measurements made at low turbulence levels without the turbulence grid. Increasing levels of free-stream turbulence are shown to reduce the magnitude of periodic disturbances produced by viscous interaction between the inlet guide vane (IGV) and rotor blade wakes.

Introduction

Unsteady flow in aeroengines is well known to influence many aspects of performance. Despite this, unsteady effects are not generally considered in current design methods due to the high level of complexity involved. There is now a large research effort to improve understanding of unsteady flow phenomena. One common source of unsteadiness in multi-stage compressors is the periodic disturbance produced by the viscous wakes of upstream blade rows. The relative flow past a blade leaves a wake region that is characterised by high levels of turbulence and lower relative velocity. These wakes are convected downstream to the next blade row, where they are chopped into segments as they pass through the blade passages. The velocity variation across the passages due to the circulation around the blades causes a rotation of the wake segments, as earlier described by Smith [6]. These segments leave the passages and then interact with the wakes shed from the blade row causing their dispersion. A typical wake dispersion pattern is shown in figure 1.

Low speed single-stage research compressors are commonly used for making detailed studies of many flow phenomena. The flow conditions are generally different from multi-stage industrial turbomachinery which operate at higher levels of turbulence due to the mixing of wakes over a large number of blade rows. Industrial machines also operate at high speeds with significant compressibility effects, a difficult environment for taking measurements. Experience has shown that many types of flow phenomena remain the same in nature in both types of machine. Consequently, low speed single stage compressors remain a widely used research tool.

Earlier studies of wake-wake interaction in the low speed research compressor at the University of Tasmania were conducted by Lockhart and Walker [3], who took hot-wire anemometer measurements of the flow in the rotor-stator axial gap. They observed that the rotor wake decay varied with circumferential position and proposed this was due to an interaction with wakes from the IGV blade row. They presented a model for the wake dispersion process similar to that shown in figure 1.

Some years later, Walker et al. [8] and Walker et al. [9] took more detailed hot-wire measurements in the rotor-stator axial gap using high speed data acquisition. The measurements were processed to calculate ensemble-averaged velocity and turbulence level. The results showed an accumulation of rotor-wake fluid on the suction side of the IGV wakes due to a restriction of the rotor wake relative flow by viscous interactions with the adjacent IGV wake segments. Results taken for different load cases showed that the rotor wake segments were turned by a larger amount as the level of loading was increased. Similar processes occur in axial turbines but the higher blade loadings produce a much larger distortion of wakes from upstream rows.

Recently Chow et al. [1] used particle image velocimetry (PIV) to study the wake-wake interaction in a 2-stage axial turbomachine. They observed regions where rotor wake fluid was collected into turbulent hot spots. They also observed a significant distortion or kinking of the rotor wakes. These observations indicate a greater amount of wake-wake interaction than has been found in the other research described here. This could be explained by their different turbomachine geometry.

The objective of the current research is to study how the wake-wake interaction process is influenced by increased levels of free-stream turbulence, such as those found in multi-stage turbomachinery.

Experimental Details

Research Compressor

Air enters the compressor radially through a cylindrical inlet 2.13m in diameter and 0.61m wide. The flow passes through a 6.25:1 contraction, where it turned 90° to the axial direction. The compressor has three blade rows: inlet guide vanes (IGV), rotor, and stator as shown in figure 1. The stationary blade rows both have 38 blades and the rotor has 37 blades, giving space/chord ratios of 0.99 and 1.02 respectively. The blade profiles were based on a British C4 section with a constant chord length of 76.2mm and an aspect ratio of 3.0. The blade profiles were stacked about a radial axis to achieve free-vortex flow and 50% reaction at mid-blade height at design flow conditions. The test section annulus is constant in area with hub and casing diameters of 0.69m and 1.14m respectively. The flow passes through an annular diffuser before discharging through a cylindrical throttle at exit. The throttle can be automatically adjusted to achieve the desired compressor load. The rotor is directly coupled to a 30kW DC motor via a long shaft. The speed is controlled by an analogue feedback loop with a computer controlled reference voltage. The speed control for a fixed set point was within $\pm 0.1 RPM$.

Instruments are inserted into the test section through an axial slot in the casing wall. A probe traversing rig on the outside allows accurate positioning in axial and radial directions.

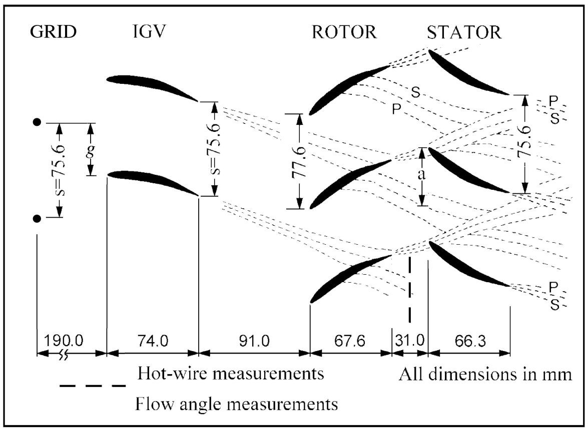

Figure 1: Cross section of the research compressor showing the mid-passage blade row configuration and a typical instantaneous wake-dispersion pattern: S = suction side P = pressure side.

The IGV and stator blade rows are held in movable rings which allow circumferential traversing over 2 blade pitches via stepper drives. This enables the stationary blade rows to be aligned circumferentially relative to each other and the fixed turbulence grid as indicated by the variables (a) and (g) shown in figure 1.

The background turbulence level of the research compressor was raised using a turbulence generating grid similar to that of Place et al. [4], who mainly focused on measuring machine performance and turbulence characteristics. The design goal was 4% turbulence intensity at entry to the stator row, which is typical of multi-stage machine operation. This was achieved by installing a turbulence grid at the start of the test section as shown in figure 1. The grid consisted of 38 radial rods of 7.94mm diameter, each spanning between rings fixed to the hub and casing. A constraint was placed on the maximum pressure loss so that the full range of test load cases could be reached. The number of rods was made equal to the number of blades in the stationary rows so that every blade in a stationary row would experience the same disturbance field. The selection of rod diameter and the grid position was primarily based on the data given in Roach [5], assuming isotropic decay of turbulence in a zero pressure gradient over an estimated mean flow path length between the turbulence grid and stator row.

Measurement Techniques

The compressor was operated at a constant blade Reynolds number ($Re = U_{mb}c/\nu = 120000$) based on mid-blade rotor speed (U_{mb}) and blade chord (c). Compressor load was controlled by setting the flow coefficient ($\phi = u_{ref}/U_{mb}$), where the reference flow speed (u_{ref}) was measured by a pitot-static tube located upstream of the test section. Ring tappings on the intake contraction were calibrated prior to installation of the turbulence grid to measure the compressor flow coefficient. The flow coefficient was set to a medium load condition ($\phi = 0.675$) during the beginning of each test and was not adjusted during testing.

The hot-wire measurements were made using a single wire Dantec 55P05 probe with sensor aligned in the radial direction. The probe support was rigidly fixed between two stator blades with the wire position in the centre of the rotor-stator gap at mid-blade height. A circumferential traverse was completed by moving the stator row and probe over one whole blade pitch keeping the IGV row fixed. The hot-wire probe was operated with a TSI IFA100 constant temperature anemometer. The frequency response of the system was estimated using a square wave test to be greater than 70kHz. The anemometer voltage was offset, amplified and low pass filtered at 20kHz before data acquisition at 50kHz. The offset and gain settings were optimised to maximise the signal range for input to the data acquisition card. Data were recorded on a Pentium II computer with an United Electronic Industries WIN30DS card. The sampling process was triggered once per revolution by a pulse from an encoder attached to the rotor shaft. Measurements were taken at 32 circumferential steps across a blade passage. In each position 512 data traces were recorded, each containing 1024 samples. This corresponds to approximately 6 wake passing periods.

The probe was calibrated using an in-situ method developed by Solomon [7]. In this method a local velocity coefficient U/U_{mb} was measured with a pre-calibrated three hole probe over a range of rotor speeds. A direct calibration was made by replacing the three hole probe with the hot-wire probe and repeating the process. Solomon [7] also investigated calibrating the probe in a different wind tunnel and then re-assembling it in the compressor for measurement. However this was found to introduce large errors caused by changes in lead contact resistances. The in-situ method eliminated this requirement and was found to be fast and repeatable.

Data Analysis

The hot-wire traces were processed using the ensemble averaging technique detailed in Evans [2]. Walker et al. [8] and Walker et al. [9] later adapted and refined this method in their research. A brief summary of their method follows.

Instantaneous velocity is commonly expressed in terms of a time mean $\overline{u}$ and associated fluctuating component u'. The flow under examination has strong periodic events and can also be defined in terms of an ensemble-averaged velocity $\langle u \rangle$ and fluctuating component u''. This may be expressed as

$$u = \overline{u} + u' = \langle u \rangle + u'' \tag{1}$$

The ensemble-averaged velocity field observed by a stationary probe downstream of the rotor is circumferentially periodic with a wavelength equal to the IGV pitch. It retains this periodicity through the stator due to the equal numbers of IGV and stator blades. It may be calculated by phase lock averaging a sufficiently large number of records N for each time instant t_i. This is expressed by

$$\langle u \rangle (t_i) = \frac{1}{N} \sum_{k=1}^{N} \{u(t_i)\}_k \tag{2}$$

The periodic unsteadiness is evaluated over an integral number of blade-passing periods and non-dimensionalised by the local free-stream velocity U. This is expressed by

$$\tilde{T}u = (\langle u \rangle - \overline{u})_{rms}/U \tag{3}$$

The true random unsteadiness is given by

$$Tu = u''_{rms}/U \tag{4}$$

The total turbulence level or overall unsteadiness is given by

$$Tu_D = u'_{rms}/U \tag{5}$$

Assuming the periodic and random turbulence levels are statistically independent they may be related by

$$Tu_D^2 = \tilde{T}u^2 + Tu^2 \tag{6}$$

Results and Discussion

Figure 2 shows processed results of the hot-wire measurements in the rotor-stator gap for the test cases with and without the turbulence grid. The shaded contour plots show ensemble-averaged velocity $< u >$, non-dimensionalised by pitchwise averaged time mean velocity $\overline{u}_s$. The line contours show ensemble-averaged disturbance level $< Tu >$ in 1% intervals. The vertical axis (w/s) is circumferential position (w) divided by the rotor blade pitch (s). Time is shown on the horizontal axis, non-dimensionalised by the rotor passing period. This convention shows the earliest measurements on the right ($t^* = 0$). The results have been replotted over a second passage by assuming the flow is periodic in the pitchwise direction. The plot represents the instantaneous view of the unsteady flow field on a cylindrical surface at mid-span radius, which would result from the flow convecting unaltered from the measuring station with zero whirl.

The rotor wakes are clearly defined by bands of high turbulence level running diagonally across the plots. These are diagonal due to the changing probe position relative to the fixed rotor position where triggering starts. The IGV wakes are shown by the horizontal segments with slightly higher than average turbulence level in the passage. This contrasts with the dispersion pattern shown in figure 1 because the whirl component of velocity has not been included. The IGV wakes are hardly visible in the high turbulence case, indicating that the elevated free-stream turbulence has accelerated their mixing out.

The contours of ensemble-averaged velocity provide further detail of the wake-wake interaction processes. The rotor wakes are clearly identified by bands of low level velocity which correspond well with the bands of high turbulence. The IGV segments have lower velocity than the mean velocity in the passage. These zones also correlate well with the contours of turbulence. At the intersection of wake streets, the lower energy fluid of the rotor wake accumulates near the suction surface side of the IGV wake. This leads to circumferential variations in the rotor wake thickness and local regions of high turbulence and lower velocity. The test case with the turbulence grid shows significantly reduced interaction. The rotor wakes fluctuate little in thickness and turbulence intensity. The flow in the passage is also much more uniform than in the case without the grid. This suggests that the higher level of free-stream turbulence has mixed out the IGV wakes. A reduction in periodic flow field at entry to the rotor may also alter the wake shedding process and contribute to a more uniform rotor wake. In particular, there should be smaller fluctuations in rotor blade trailing edge boundary layer thickness due to reduced unsteadiness in the transition process on the blade surface.

The line graphs on the left hand side of figure 2 show time mean values of turbulence level and velocity against circumferential position. Apparent turbulence level Tu_D is shown with the periodic component T$\tilde{u}$ and random component Tu. In both cases, the random component is greater in the IGV wake. The case with higher turbulence shows only a very slight increase at the location of the IGV wake. Significant periodicity occurs at the position where the low energy rotor wake fluid has collected. This also corresponds to a minimum of time mean velocity. The periodic unsteadiness peaks are essentially absent with the turbulence grid installed.

The turbulence intensity between blade wakes was only slightly lower than the design value of 4%. The small deviation from design was most likely due to the neglected effects of changing stream velocity.

Conclusions

A study of increasing inlet turbulence level in a 1.5 stage axial compressor has shown a strong influence on wake-wake interactions. Hot-wire measurements taken in the rotor stator gap were used to calculate ensemble-averaged velocity and turbulence. At low levels of inlet turbulence the results showed strong periodic fluctuations in rotor wake thickness and ensemble-averaged velocity. At high levels of inlet turbulence the periodic fluctuations were significantly reduced. This indicates that viscous interaction processes will be much smaller in magnitude for embedded blade rows in a multi-stage axial turbomachine.

Acknowledgments

The financial support of Rolls-Royce plc is gratefully acknowledged.

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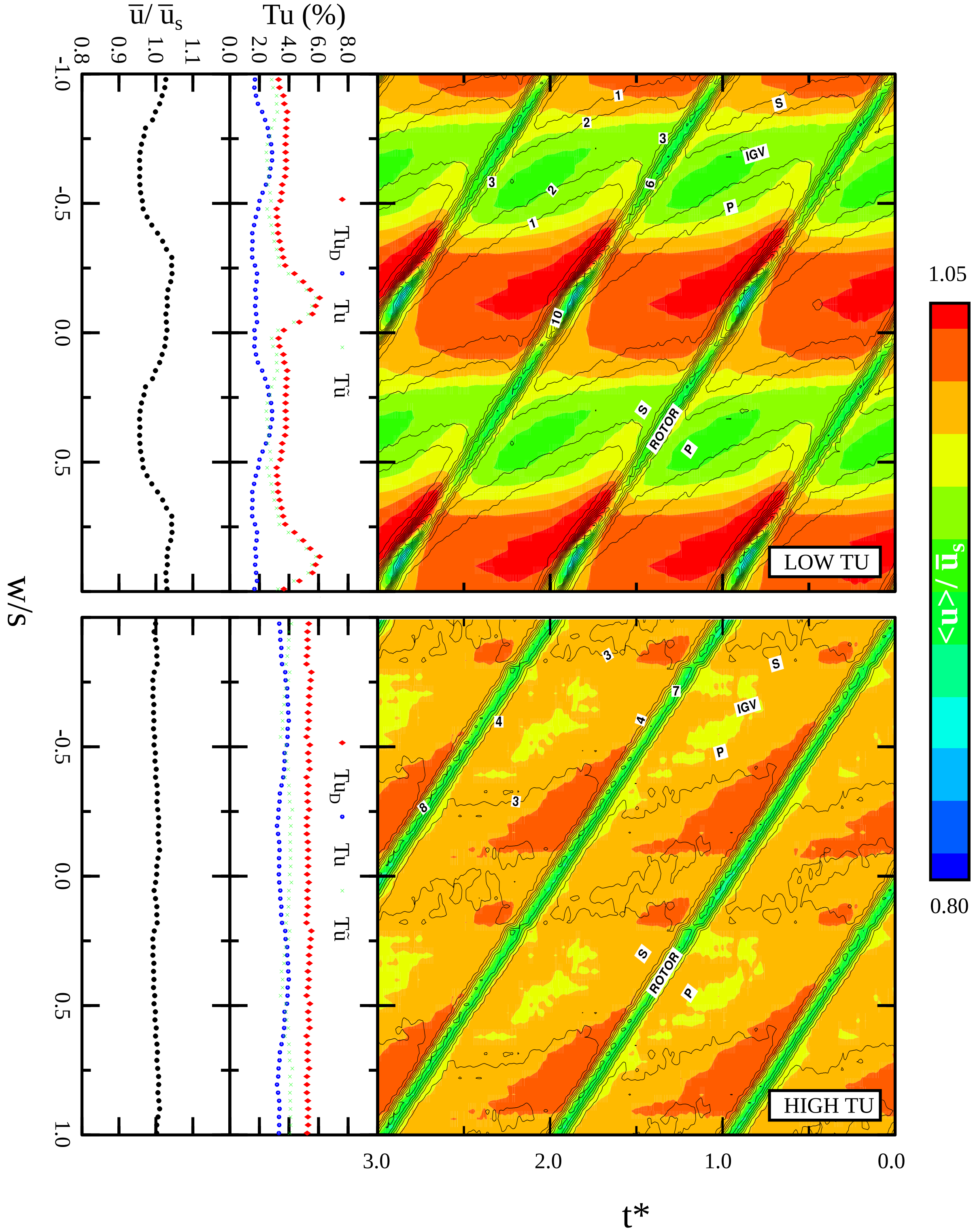

Figure 2: Variation of ensemble-averaged velocity with turbulence level for low (top) and high (bottom) levels of free-stream turbulence. Hot-wire measurements taken in the rotor-stator passage at with a flow coefficient $\phi = 0.675$ and blade Reynolds number $Re = 120000$. Shaded contours show ensemble-averaged velocity $< u > /\overline{u}_s$. Line contours show ensemble-averaged turbulence level $< tu >$ in 1% intervals.